



RESEARCH DEPARTMENT

REPORT

**An investigation into the mechanism
of sound-energy absorption in a
low-frequency modular absorber**

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AN INVESTIGATION INTO THE MECHANISM OF SOUND-ENERGY
ABSORPTION IN A LOW-FREQUENCY MODULAR ABSORBER

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Summary

An investigation into the behaviour of a widely used type of modular, low-frequency acoustic treatment is described. A region of spurious sound-energy absorption in the middle part of the audio frequency range, which was originally acceptable, has become more significant in units made since the prototypes were tested in 1969. This has caused excessive absorption in treated rooms in this range of frequencies.

In an effort to identify the cause of this spurious absorption, a theoretical study was carried out, based on the supposed behaviour of this absorber. This analysis failed to identify the cause, showing that the supposed behaviour was erroneous.

Experimental work on a number of modifications to the original design showed that the spurious absorption was caused by damped vibrations of the main structure of the modules. It also showed that the wanted low-frequency absorption resulted from mechanical vibrations of the front-panel, rather than from an acoustic ('Helmholtz') resonance, as had previously been thought.

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1. Introduction

When designing a building it is usually the case that, where acoustic performance is important, an enclosed space has to be acoustically treated with some form of sound absorbing material or structure in order to achieve the required characteristics. The earliest written accounts of such deliberate acoustic treatment are in the 'Ten Books on Architecture' of the Roman, Vitruvius, written in 20 B.C., which describe the use of resonators to modify the reverberation time in Greek amphitheatres¹. However it was not until the early part of the 20th century that the subject of acoustic room treatment was placed on a firm, scientific basis by Sabine's work at Harvard.² It was Sabine who formally developed the concept of reverberation time and showed that there is an optimum value which depends on the size of the enclosure and on the use to which the enclosure is to be put. He also showed that the value of the reverberation time can be modified by suitable treatment of the surfaces of the enclosure. This report discusses the use, in the BBC, of one particular type of acoustic treatment which has been in use for about 12 years. Over this period of time, some variation has caused what was originally a tolerable departure from the required characteristic, to become a serious problem. An effort was made to calculate the theoretical performance of this acoustic treatment and also to measure experimentally the effects of a number of changes in the construction. By these means, it was hoped that the reason for the anomalous behaviour could be identified, thus showing which detail of the construction had changed since the prototype was developed.

2. The acoustic treatment of areas used for broadcasting.

The requirement for acoustic treatment is especially evident in broadcasting studios and control rooms (cubicles) where the reverberation time must be controlled within close tolerances. As an example, the required reverberation time for a radio 'talks' studio, where the normal source of the sound is the human voice, is around 0.3 seconds. This value should be maintained to within about

$\pm 10\%$ over the whole range of audio frequencies although, for talks studios only, an increase beyond this at very low frequencies is regarded as permissible³. Any significant departure from this ideal would probably result in difficulties being experienced both by the users of the studio and the operator trying to achieve a satisfactory sound quality from the microphone output. Other types of studios and control rooms differ only in detail, the principles involved in all such areas are the same.

In most of such rooms, the reverberation time without any acoustic treatment is much too long and it is almost always necessary to reduce it by the addition of sound absorptive treatment to the basic structure. In rooms with relatively lightweight boundaries, which can absorb low-frequency sound energy, the reverberation time can fall to a lower value at low frequencies. Further, in large rooms, the effect of air absorption can become significant at high frequencies and can cause a reduction of the reverberation time at the high-frequency end of the spectrum. Thus, the acoustic treatment should provide a uniformly efficient sound-absorbing mechanism at most frequencies, with some more adjustable characteristics at both the low and high-frequency ends of the audio frequency spectrum.

The usual method of specifying the performance of a sound-absorbing treatment is to define the absorption coefficient at any frequency as the ratio of the absorbed sound-energy to the incident sound-energy. Alternatively, the equivalent concept of the ratio between the actual surface area of the test material and the area required to produce the same energy absorption of a material which absorbs all of the incident energy can be used. Numerically, these two interpretations are the same and the resultant absorption coefficient, α , is usually less than unity.

Fig. 1 illustrates the variation, with frequency, of idealised absorption coefficients required for the treatment of broadcasting studios and cubicles of various types. The requirements for the absorption coefficient to be as high as possible arise because higher values reduce the total area of treatment needed to obtain the required reverberation time; it is sometimes difficult to find

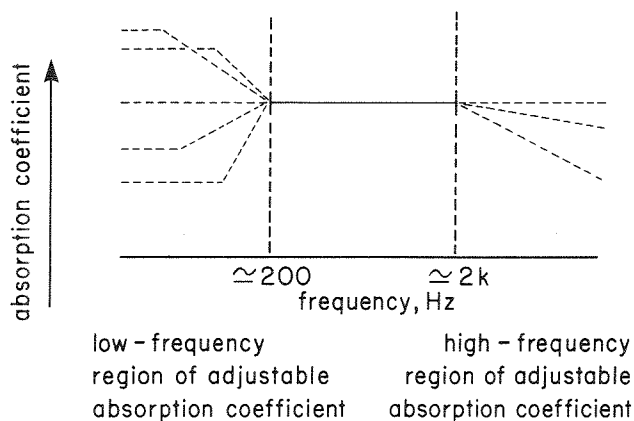


Fig. 1 - Absorption coefficient characteristic required for the treatment of a range of studios and control rooms (cubicles).

sufficient space for the necessary treatment, especially in sound control rooms where the apparatus can occupy a significant fraction of the wall area.

In practice, such ideal absorbers cannot be achieved realistically*, but various forms of acoustic treatment have been developed to treat parts of the frequency spectrum separately. One form consists of a broad range of 'modular absorbers' having a superficial area of approximately 0.58 m x 0.58 m and three different depths or thicknesses in the range 0.107 to 0.215 m. The construction of these modules consists of a front panel usually 3 mm thick perforated hardboard, a layer of 30 mm mineral-wool, an airspace and finally, the back of the unit, made of 3 mm or 6 mm unperforated hardboard. A solid wood or thick plywood frame forms the perimeter of the module and supports the various panels around their edges. The enclosed airspace is usually subdivided into about 50 spaces by cardboard or hardboard partitions parallel to the side walls. In some cases, the perforated front-panel is positioned behind the mineral-wool, which in turn is held in place by a front covering of very open wire mesh.

A wide range of characteristics can be achieved by variations in this construction and by varying the open-area of the perforated hardboard and the density of the mineral-wool. A summary of the different types is given in Appendix 1. One of the advantages of such a modular system is that the

* It is probable that a 1 m thick layer of mineral-wool placed on the wall would achieve a uniform absorption coefficient, at least at frequencies higher than 100 Hz. However, this is not generally a feasible or cheap solution, and would in any case be very difficult to modify at specific frequencies to allow for room variations.

uniform size facilitates the initial layout and enables subsequent remedial modifications to be achieved simply by altering the front panels and the contents of the boxes. This system was first used for temporary local radio premises in 1967, the modules being demountable for subsequent removal to permanent premises; they have been used widely in the BBC ever since for all types of studios and control rooms.

In the case of radio studios, the types most commonly used in the BBC have been the A3 (wideband) and the A2 (low-frequency) modules. The A3, which has a 20% perforated hardboard front-panel and a 30 mm mineral-wool layer over a 144 mm airspace, should give a uniformly high absorption coefficient from about 400 Hz to about 2 kHz. At frequencies higher than 2 kHz the absorption coefficient should gradually become smaller as a result of the perforated hardboard front-panel behaving as a low-pass filter. This allows for air-absorption in moderate-size rooms. In practice, for reasons which are discussed later, the high absorption coefficient extends to frequencies lower than 400 Hz. A typical absorption coefficient characteristic for the A3 is shown in Fig. 2.

The A2 module is identical to the A3 in all respects except that the front-panel open-area ratio is 0.5%. Theoretically, this should result in a resonant absorber with an absorption coefficient peaking at about 100 – 120 Hz, as shown in Fig. 3. When used to complement the A3, and with the correct ratio of A2 to A3 modules, a reasonably uniform reverberation time characteristic can usually be achieved over the required audio frequency spectrum.

However, it became evident that such a method of treatment, whilst being satisfactory at

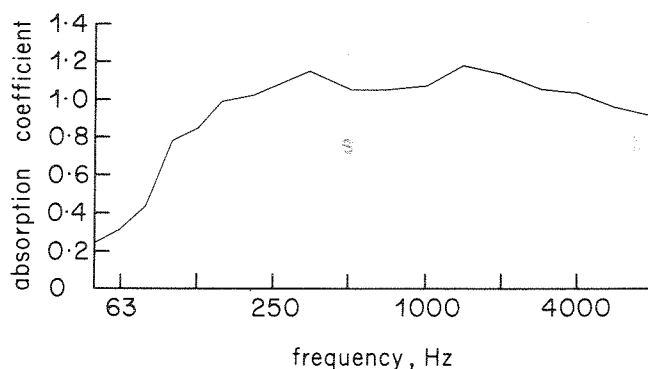


Fig. 2 - Typical absorption coefficient characteristic for an A3, wideband modular absorber.

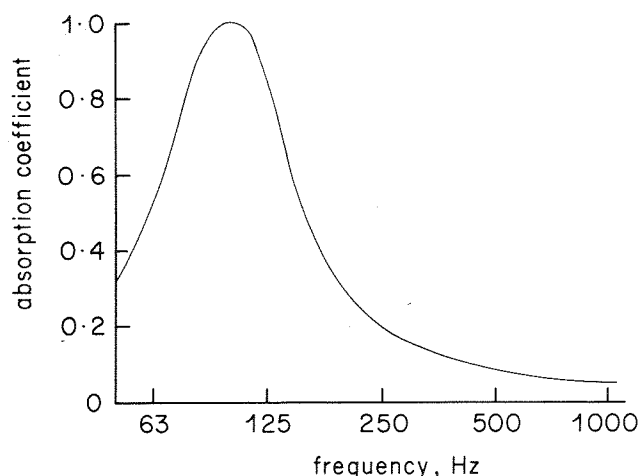


Fig. 3 - Hypothetical absorption coefficient characteristic for a resonant low-frequency modular absorber.

both low and high frequencies, resulted in excessive absorption in the region around 450 Hz, extending to about one half octave on either side of this frequency. Subsequent investigations showed that the A2 absorbers were not showing the required characteristics, and that there was a large spurious peak in the absorption coefficient characteristic at about 400 Hz. Fig. 4, characteristic (a) shows a typical measured absorption coefficient as a function of frequency for such an absorber, compared with characteristic (b) a measurement of a sample set of

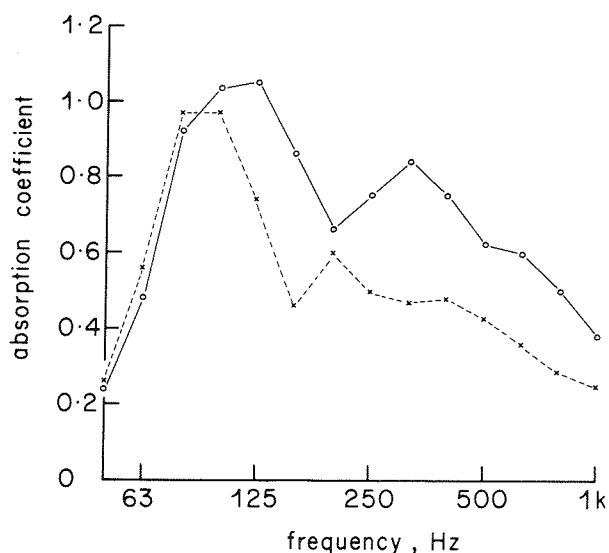


Fig. 4 - Measured absorption coefficient characteristic for a real A2 modular absorber.

- (a) —○— typical current measurement
(b) - - - x - - - original measurement on the prototype, 1969.

the original units, measured in 1969. It is evident that, during this period, there has been a significant increase in the absorption coefficient at all frequencies above 100 Hz. Any calculations of treatment quantities based on the original figures would thus be seriously in error, particularly around 400 Hz. Although it seems that the spurious absorption has certainly increased since the A2 modular absorbers were first measured, it is evident from characteristic (b) that the original measurements show a similar but less pronounced effect. It was therefore necessary to find the fundamental cause of this spurious absorption before any properly based alterations to the design could be carried out.

3. Theoretical calculation of the absorption coefficient of an A2 modular absorber.

A cross section of an A2 modular absorber, including the corner detail, is shown in Fig. 5. A complete mathematical description of its acoustic performance, including the adjacent surroundings and the volume of the room in which it is placed is exceedingly difficult. However, several aspects of its behaviour can be considered individually in order to obtain an idea of the magnitude of the total absorption coefficient. The model to be considered therefore incorporates some simplifications. First, only sound incident along the direction of the normal to the surface will be considered. This is reasonably justifiable because the airspace behind the front panel is subdivided into small compartments which reduces the interaction between adjacent parts of the front surface. Further, these individual parts of the front surface are sufficiently small, at the frequencies of interest,

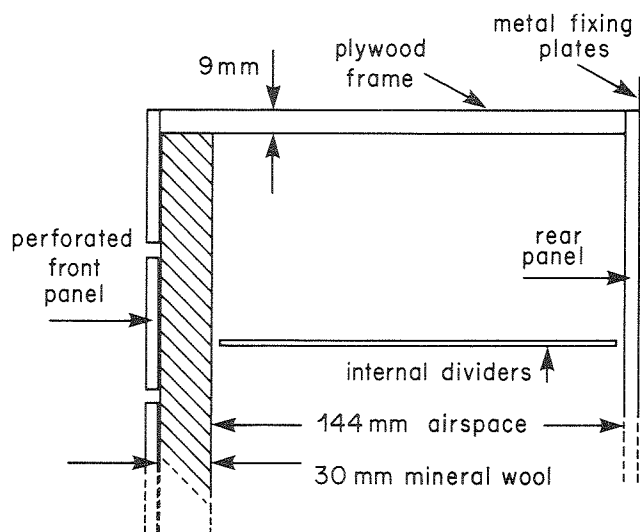


Fig. 5 - Cross-section of A2 modular absorber.

to behave non-directionally because of diffraction effects. Second, the front and rear panels are considered to be immovable, that is, they have sufficient mass or stiffness to be unaffected by the incident sound. Later, this will be shown to be an invalid assumption, thus demonstrating that the hitherto assumed behaviour of this modular absorber is not entirely correct.

Thus, the model of the absorber consists of a rigid, perforated front-panel of open area 0.5% (3 mm diameter holes spaced 38 mm apart on a square grid) backed by a 30 mm low-density mineral-wool layer, a 144 mm airspace and, finally, a rigid unperforated back-panel.

Two mechanisms for sound absorption exists for this structure. The first is a 'Helmholtz' type of resonator formed from the mass of the air in one of the holes, together with the compliance of the air contained in that particular holes' share of the backing airspace. The mineral-wool layer damps this resonator, thus providing energy absorption. Secondly, the mineral-wool, spaced at a distance d from the rigid backing panel, will have an absorption coefficient characteristic which consists of an infinite series of regularly spaced peaks at frequencies which are odd multiples of $\lambda/4d$. This effect will, however, be modified by the series impedance of the holes in the front panel. The holes will behave as a low pass filter which will reduce this effect and hence reduce the absorption coefficient at higher frequencies.

3.1 The Helmholtz resonator

An elementary portion of the absorber consists of a longitudinal section, 38 mm x 38 mm square and about 184 mm long. Fig. 6 shows an isometric diagram of this section. The hole diameter is 3 mm and the nominal front-panel thickness is 3.2 mm.

The acoustic mass of the air contained by hole, M_A , is given by (from Ref. 4):

$$M_A = \rho_o (t + 2l'')/\pi a^2, \text{ for small } a$$

where

a = the radius of the hole ($=1.5 \times 10^{-3}$ m)

t = panel thickness ($=3.2 \times 10^{-3}$)

l'' = end correction $= 0.85a$ ($=1.275 \times 10^{-3}$ m)

ρ_o = standard air density ($=1.18 \text{ kg. m}^{-3}$)

this gives: $M_A = 960 \text{ kg. m}^{-4}$

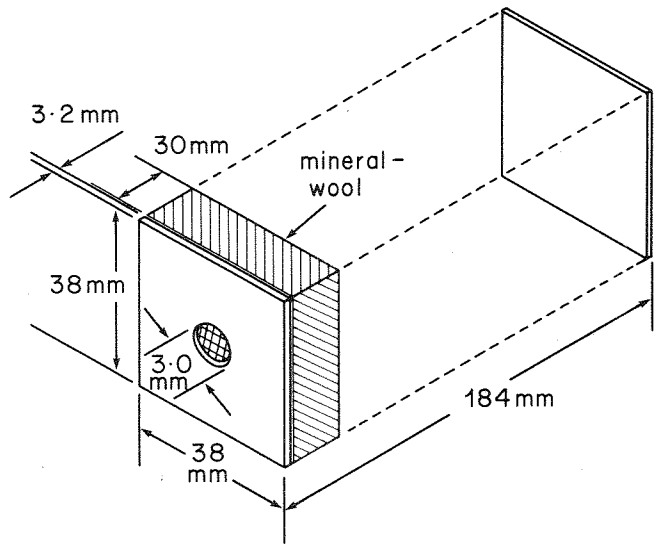


Fig. 6 - Elementary isometric section of A2 modular absorber.

The compliance of the air space behind the front-panel is that of a square section tube with 38mm sides and approximately 190 mm long, assuming adiabatic conditions in the airspace and isothermal conditions in the absorbing layer.

The effective radius a' of the equivalent cylindrical volume is given by:

$$a' = \sqrt{S/\pi}$$

where S is the cross-sectional area ($=b^2$)

and b is the spacing between the holes.

Hence, $a' = 2.14 \times 10^{-2} \text{ m}^2$

The acoustic input impedance of such an enclosed volume is given by (from Ref. 5):

$$Z_A = -j / (\omega V / (\rho_o c^2)) + (j \omega l' \rho_o) / (3 \pi a'^2)$$

where Z_A is the complex acoustic impedance

V is the volume of the airspace ($=2.73 \times 10^{-4} \text{ m}^3$)

c is the velocity of sound in air ($=344 \text{ m.s}^{-1}$)

$\omega (=2\pi f)$ is the frequency in radians. s^{-1}

l' is the length of the airspace in the direction of sound propagation.

The first term incorporates the acoustic compliance, C_A , where

$$C_A = V/(\rho_o c^2)$$

$$= 1.957 \times 10^{-9} \text{ m}^5 \cdot \text{N}^{-1}$$

At 100 Hz, this is equivalent to an acoustic compliance reactance of 8×10^5 acoustic ohms. The second term is of the form of an acoustic mass reactance and is a correction for the shape of the enclosed volume, the primary compliance term being independent of the shape. After substitution, the second term gives 3.2×10^4 acoustic ohms at 100 Hz. This is equal to 4% of the impedance given by the first term and can therefore be neglected.

Resonance occurs when the sum of the mass reactance of the air contained by the hole in the front-panel and the compliance reactance of the enclosed airspace is equal to zero. That is:

$$j\omega M_A - j/(\omega C_A) = 0$$

which gives,

$$\omega = 1/(\sqrt{M_A C_A})$$

$$= 729.6 \text{ radians} \cdot \text{s}^{-1}$$

$$= 116 \text{ Hz}$$

This is, theoretically, the desired mode of operation and has a resonance at almost exactly the desired frequency to make a good low-frequency absorber. However, experimental evidence described later in Section 4.4.2, shows that it is doubtful whether the absorber behaves entirely in this manner, even at the frequency at which it appears to work well in practice.

3.2 Absorption at other than the Helmholtz resonance frequency

In order to obtain the absorption coefficient at a number of different frequencies it is advantageous to derive an equivalent circuit which represents the acoustic parameters as electrical components. The network-solving techniques which are a familiar part of electrical circuit theory can then be applied to the equivalent of the acoustic network representing the absorber. Many sets of equivalence relationships are possible, but the one which will be used here equates electrical current to volume velocity and electrical voltage to acoustic pressure. With this convention, mass is represented by inductance, compliance by capacitance and energy loss by resistance.

For the model absorber as previously defined, the equivalent circuit can be derived by inspection

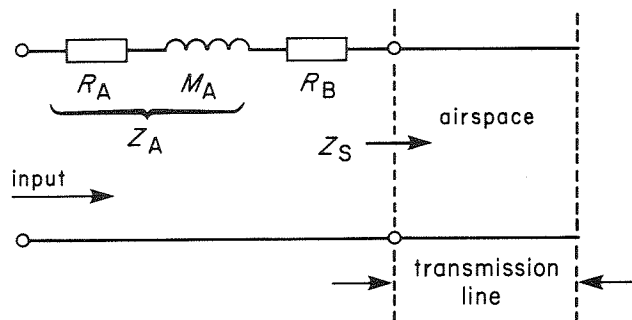


Fig. 7 - Equivalent circuit of the lumped acoustic parameters associated with one of the holes of the perforated front-panel.

and is shown in Fig. 7 for one of the holes in the perforated front-panel. R_A and M_A represent the resistive and inductive components of the perforated front-panel and R_B represents the resistance of the mineral-wool layer. The airspace is represented by a transmission line of length $\lambda/4$ at 450 Hz, where λ is the wavelength of the sound wave. The back-panel is represented by an open-circuit transmission-line termination.

The terms R_A , M_A and Z_S can be readily calculated. Fig. 8 shows a schematic representation of the front panel.

The effective series impedance, Z_A , of one hole can be derived (from Ref. 6):

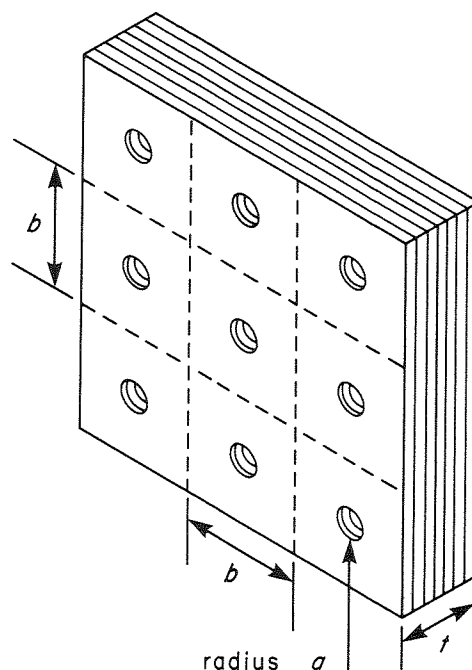


Fig. 8 - Schematic representation of the perforated front-panel.

$$Z_A = R_A + j\omega M_A$$

where $R_A = (\rho_o / \pi a^2) (\sqrt{2\omega\mu} (t/a + 2(1 - A_h/A_b)))$

$$M_A = (\rho_o / \pi a^2) (t + 1.7a (1 - a/b))$$

and $\mu = \text{kinematic viscosity} (= 1.56 \times 10^{-5} \text{ m}^2 \text{ s}^{-1})$

$A_h = \text{cross sectional area of the hole}$
 $(= 7.07 \times 10^{-6} \text{ m}^2)$

$A_b = \text{area of front panel associated with}$
 $\text{one hole } (= b^2 = 1.44 \times 10^{-3} \text{ m}^2)$

and the other terms as previously defined.

These expressions are valid over a range

$$0.01\sqrt{f} < a < 10/f$$

i.e. from 45 Hz to 6.7 kHz.

By substitution:

$$M_A = 938.9 \text{ kg. m}^{-4}$$

and $R_A = 3829\sqrt{\omega} \text{ acoustic ohms}$

This value for M_A is slightly smaller than that previously calculated for the Helmholtz resonator, due to the correction term $(1 - a/b)$ which accounts for the proximity effect of the holes. However, the difference is sufficiently small to be irrelevant to the purposes of this work.

The term R_B is difficult to derive theoretically as it involves a two-dimensional solution to the wave equation for the mineral-wool layer but a value sufficiently accurate for the present purpose can be found from the measured effect of the absorbing layer on the Q-factor of the resonant system.

At 120 Hz, substituting in the above expressions for R_A and M_A

$$Z_A \approx 105 \times 10^3 + j 708 \times 10^3$$

and therefore, $Q (= \omega M_A / R_A) = 6.7$ at 120 Hz, if the front panel alone was to be made resonant by the addition of a perfect compliance. The actual values of Q as measured are about 3.4. Thus, the effect of the mineral-wool layer is to insert an additional resistive component approximately equal to the loss component of the front panel at 120 Hz, that is, 105×10^3 acoustic ohms/per hole. The assumption is made that the acoustic resistance of the mineral-wool layer is independent of frequency. This is true of this type of

material to within a factor of 2 : 1 over the frequency range 0 – 4 kHz.

The equivalent circuit of the whole model absorber is therefore as shown in Fig. 9, where the individual impedances have been divided by the total number of holes in parallel (≈ 220).

At low frequencies, the input impedance of the airspace is almost purely capacitive and it is this value of capacitance (compliance) which resonates with the mass of the sir in the hole at the Holmholtz resonant frequency.

For the purposes of this work, it will not be necessary to calculate the complex input impedance of the airspace. As the object is to determine the maximum absorption coefficient of the absorber, it will be adequate to consider only those frequencies at which there is a local peak in the absorption

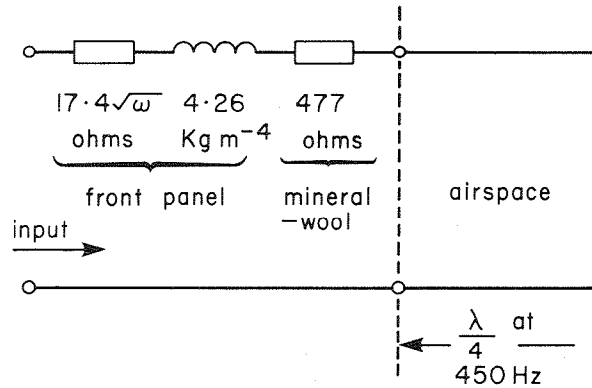


Fig. 9 - Equivalent circuit of the whole model absorber.

coefficient. At these frequencies, the input impedance of the transmission line is zero, thus giving maximum volume velocity in the absorbing layer. These frequencies are 450 Hz and odd integer multiples thereof, 1350 Hz, 2250 Hz, 3150 Hz and so on.

The input impedance of the whole panel can be readily calculated at these frequencies. Table 1 gives the calculated input impedances.

In addition, there is another series of very narrow peaks in the absorption coefficient, at frequencies where the input impedance of the airspace is capacitive and of the correct magnitude to cancel the remaining inductive components of the system. The lowest of these frequencies is 915 Hz and the second and third are 1,825 Hz and 2,720 Hz, respectively. Although the absorption coefficients for these frequencies have been calculated and will be given for completeness, they

Table 1

Calculated input impedance of model absorber panel

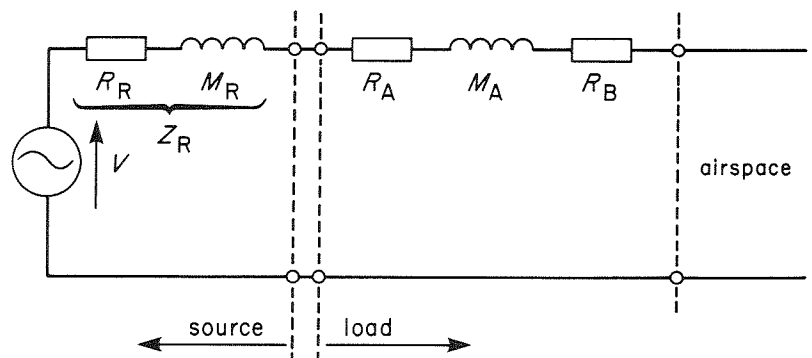
Frequency		R_A	$R_A + R_B$	$j \omega M_A$
Hz	Radian s^{-1}	Acoustic ohms		
450	2 827	925	1 402	12×10^3
1 350	8 482	1 602	2 079	36×10^3
2 250	14 137	2 068	2 545	60×10^3
3 150	19 792	2 448	2 925	84×10^3
4 050	25 447	2 776	3 253	108×10^3
4 950	31 102	3 068	3 546	132×10^3

will not be considered in detail as this mechanism cannot be responsible for the excess absorption at 450 Hz, which is the main subject of this report.

3.3 Absorption of sound energy

The absorber equivalent circuit can be considered as a lossy load connected to a source of voltage representing the sound pressure. The impedance of this source is, by reciprocity⁷ equal to the radiation impedance of the absorber. This is illustrated in Fig. 10 where Z_R is the complex radiation impedance. The absorption coefficient is defined as the ratio of the actual power absorbed by the load to the power which would be absorbed if the load impedance were matched for maximum power transfer to the source impedance. In most theoretical treatments of absorbers, the source impedance and the load which gives the maximum power transfer are taken to be equal to the free-space acoustic impedance, $\rho_0 c$, which is purely resistive and has a value of 407 acoustic ohms for standard conditions of temperature and pressure.

Fig. 10 - Equivalent circuit of the whole model absorber, connected to a source of finite impedance.



However, this is only appropriate for cases where the absorber is effectively infinite in extent. Where the absorber has finite boundaries, the radiation impedance consists partly of reactive components.

A common term for an ideal absorber is the so-called 'open-window', but neither is this appropriate in the cases where the radiation impedance is complex, that is, where the absorber is not very large in comparison with the wavelength of the sound energy. It is equivalent to terminating the source impedance by an identical load impedance.

In these cases, the reference condition for maximum sound power absorption should be with a load impedance which is the complex conjugate of the source impedance.

It can be shown that the absorption coefficient, α , as defined above is given by (see Appendix 2):

$$\alpha = (4 R_R (R_A + R_B)) / (R^2 + X^2) \quad (1)$$

where R_R = resistive part of the radiation impedance

$R_A + R_B$ = resistive part of the load impedance

R = total circuit resistance

X = total circuit reactance

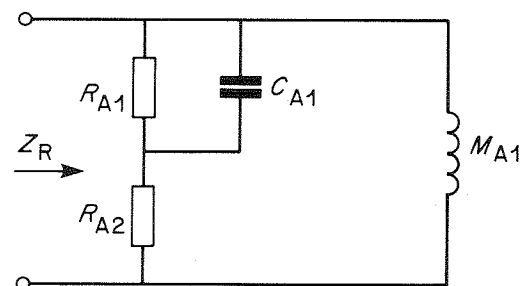


Fig. 11 - Approximate equivalent circuit representing the radiation impedance.

3.4 Radiation impedance

An approximate equivalent circuit of the radiation impedance, Z_R , of one side of a circular piston mounted in an infinite plane is given in Fig. 11 (from Ref. 8) where:

$$R_{A1} = 0.1404 \rho_o c / A^2$$

$$R_{A2} = \rho_o c / (\pi A^2)$$

$$C_{A1} = 5.94 A^3 / (\rho_o c^2)$$

$$M_{A1} = 0.27 \rho_o / A$$

$$c = \text{velocity of sound in air } (= 344 \text{ m.s}^{-1})$$

$$\rho_o = \text{density of air } (= 1.18 \text{ kg.m}^{-3})$$

$$A = \text{equivalent radius of a circular absorber of the same superficial area. } (= 0.3439 \text{ m})$$

Using these equations, a table can be constructed showing the radiation impedance of the absorber at the same frequencies as used in Table 1 with the addition of the Helmholtz resonant frequency (Table 2).

3.5 Calculation of absorption coefficient

Using equation 1 and the values listed in Tables 1 and 2 for the absorber input impedance and radiation impedance, the absorption coefficient at each frequency can be calculated. Table 3 gives the results of these calculations. An additional line, for 105 Hz, is included because the inductive component of the radiation impedance lowers the Helmholtz resonant frequency. Taking this into account gives a frequency of 105 Hz.

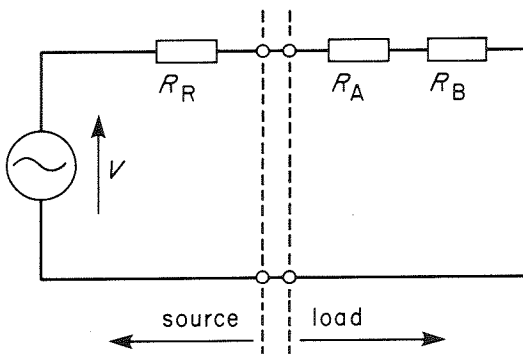


Fig. 12 - Equivalent circuit of source-absorber system at the frequencies where the sum of all circuit reactances is zero.

3.6 Other local maxima of absorption coefficient

As mentioned in Section 2.2, another series of frequencies exists at which peaks in the absorption coefficient occur. With reference to Fig. 10, the input impedance of the transmission line which terminates the absorber front structure will, over a sufficiently wide range of frequencies, take all possible values from $+j.\infty$ to $-j.\infty$. As there is no loss mechanism in the transmission line, at least in the model, there can never be a real component only a reactive component. Therefore, at some frequencies this input impedance must be capacitive and of just sufficient magnitude to cancel all of the remaining inductive reactance. The lowest of these frequencies has been calculated to be 915 Hz. Under these conditions, the equivalent circuit of the source/absorber system is as shown in Fig. 12 and the calculated absorption coefficients are given in Table 4.

At first sight, the results shown in Table 4 are extraordinary, especially as no record exists of their ever being measured in practice. However, it is obvious from consideration of the mechanism which causes them that they are very narrow peaks. A calculation of the absorption coefficient in the vicinity of the peak at 915 Hz shows that the range of frequencies over which it is significant is very small. Fig. 13 shows the results of this calculation. (Note the logarithmic scale). In any case, the normal absorption measurement technique, which will be described below, is not capable of detecting such very narrow peaks in the absorption coefficient. However, some experimental confirmation of this effect was obtained by measuring the sound pressure level inside an absorber as a function of frequency around 1 kHz. A very narrow resonance peak was found at about 900 Hz.

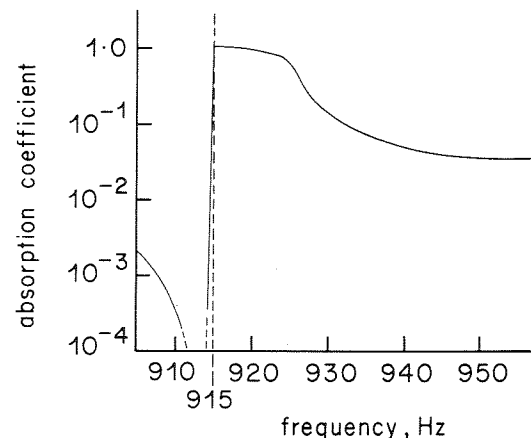


Fig. 13 - Absorption coefficient characteristic in the vicinity of 915 Hz.

Table 2

Calculated radiation impedance of the model absorber

Frequency Hz	R_{A_1}	R_{A_2}	$-j/\omega C_{A_1}$	$j\omega M_{A_1}$	Radiation impedance
	Acoustic ohms				
120	483	1 096	771	698	$304 + j\ 597$
450	”	”	206	2 617	$1\ 092 + j\ 336$
1 350	”	”	69	7 854	$1\ 102 + j\ 88.6$
2 250	”	”	41	13×10^3	$1\ 098 + j\ 51.5$
3 150	”	”	29	18×10^3	$1\ 096 + j\ 36.5$
4 050	”	”	23	24×10^3	$1\ 096 + j\ 28.3$
4 950	”	”	19	29×10^3	$1\ 096 + j\ 23.1$

Table 3

Calculated absorption coefficients at the frequencies where the transmission line length is equal to an odd number of quarter wavelengths

Frequency Hz	Radiation Impedance	Absorber Input Impedance	Absorption Coefficient
	Acoustic ohms		α
105	236 + j 546	924 – j 546	0.648
120	304 + j 597	954 + j 0	0.598
450	1 092 + j 336	1 402 + j 12 x 10 ³	38.6 x 10 ⁻³
1 350	1 102 + j 88.6	2 079 + j 36 x 10 ³	6.98 x 10 ⁻³
2 250	1 098 + j 51.5	2 545 + j 60 x 10 ³	3.09 x 10 ⁻³
3 150	1 096 + j 36.5	2 925 + j 84 x 10 ³	1.81 x 10 ⁻³
4 050	1 096 + j 28.3	3 253 + j108x 10 ³	1.22 x 10 ⁻³
4 950	1 096 + j 23.1	3 546 + j132x 10 ³	0.87 x 10 ⁻³

Table 4

Calculated absorption coefficients at the frequencies where the sum of all of reactive components is zero.

Frequency Hz	Absorption Coefficient
915	0.98
1 825	0.92
2 720	0.87
3 635	0.83
4 540	0.80

3.7 Summary of theoretical analysis

Fig. 14 shows all of the calculated peaks in the absorption coefficient (on a logarithmic scale) as defined above for the assumed model of the A2 modular absorber. Fig. 14 also shows, on the same scale, the measured absorption coefficient of a typical A2 absorber in practice (the same data plotted on a linear scale in Fig. 4). The results appear to show that the low frequency absorption at the fundamental resonant frequency can be predicted to a reasonable degree of accuracy. However, the predicted performance over the remainder of the frequency spectrum, especially around 450 Hz, is so different to the measured

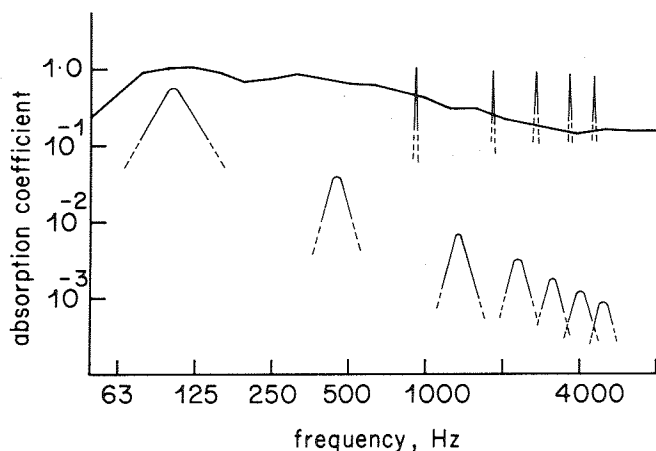


Fig. 14 - Summary of all of the calculated maxima in the absorption coefficient characteristic.

- (a) calculated local maximum
(b) typical measured characteristic.

performance that it is clear that the model is deficient in that vital aspect.

One simplification which might seem questionable is the calculation of the impedance of the mineral-wool layer, and the assumption that this impedance is independent of frequency. However, this resistive component appears in the equivalent circuit in series with the element representing the mass of the air in the hole. Even at 450 Hz., this mass reactance is so large (about 30 x larger than the resistance of the mineral-wool layer) that variations in the resistance of the mineral-wool layer are relatively insignificant. It is this mass reactance that makes the perforated front panel behave as a low-pass filter, as it was expected to do.

One very important simplification was the assumption that the front-panel, the rear-panel and the sides of the absorber were infinitely stiff or massive. An elementary calculation based on the simplest conceivable mode of vibration of the structure, that is the front-panel mass resonating as an infinite limp diaphragm with the compliance of the enclosed airspace, gave a resonant frequency of 84 Hz. In practice, the frequency would be higher than 84 Hz as the front-panel has stiffness as well as mass. This is close to the calculated Helmholtz resonance frequency and to the calculated frequency of the maximum absorption coefficient. However, a complete mathematical analysis of the modes of vibration of the whole complex structure, especially in the region of 450 Hz where the modal density would be high, is beyond the scope of this present work.

4. Experimental work

Over a number of years, a great deal of experimental work has been carried out on modular absorbers, some of which is relevant to this particular problem. In particular a series of measurements was carried out on a set of modular absorbers in an attempt to identify the mechanisms for sound absorption, especially in the region of the unwanted absorption at frequencies around 450 Hz.

In all the following results, a comparison is made between the effects of varying one particular feature at a time, all others being unaltered as far as practically possible. In practice, the variation being illustrated may not be the only one involved and so the different sets of results are not comparable with each other. Within each set of results, however, all the parameters except the one being illustrated are the same.

4.1 Measurement technique

The most realistic assessment of the likely performance of acoustic treatment in real environments is obtained by observing (in third-octave bands) the reduction in the reverberation time of an enclosure as a consequence of installing the treatment. The mean absorption coefficient can then be obtained in each frequency band in terms of the equivalent area of perfectly absorbing material. Hence the coefficient of performance of the known area of treatment can be directly calculated. Two conditions must be fulfilled before this technique can give meaningful results. First, for accuracy the change in the reverberation time caused by the introduction of the material under test must be large. This requires the untreated reverberation time to be long, such as is obtained in a reverberation chamber, and the use of a reasonably large quantity of the treatment to be tested. Secondly, in order to get a more accurate measurement of the way the treatment will behave in practice, the sound field should be reasonably diffuse, as it is likely to be in real applications. This requires the reverberation chamber to be large and arranged to give good diffusion. It has also been shown that the way in which the treatment is subdivided affects the results. To obtain comparable results the International Standards Organisation (ISO) has defined the standard volume of the reverberation chamber as 200 m^3 . The ISO has also specified that the treatment under test should have an surface area of 10 m^2 and be laid as a single panel on the floor. Tests have shown

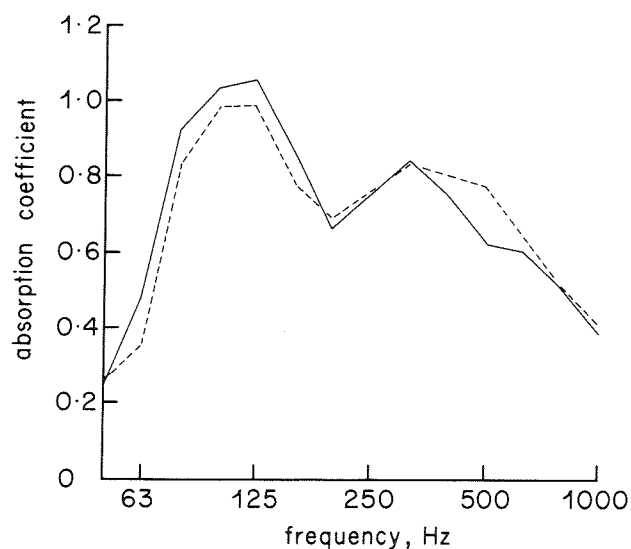


Fig. 15 - Effects of variations in the distribution of the test modules in the reverberation chamber.

- (a) 4 groups of 6 modules.
- - - (b) 12 groups of 2 modules.

that results more representative of actual conditions can be obtained by subdividing the area of treatment and it is normal practice in the BBC Research Department to divide the treatment into four equal areas applied to three walls and the floor of the reverberation chamber. In testing standard modular absorbers, it is usual to use either twenty-four or sixteen units giving areas of approximately 14.5 m^2 or 10 m^2 divided into four groups of six or four. This requirement for a large number of absorbers leads to difficulties in the experimental work as any modification to the design requires the alteration of at least sixteen units. Therefore, the modifications must be very carefully considered as time is generally not available for the testing of large numbers of different designs. Up to now, no satisfactory method has been found for obtaining the absorption coefficient characteristic for a real environment from measurements made on one or two units, although attempts to do this have been made.

4.2 Measurements based on an unmodified A2

The first aspect to be investigated was whether the measurement technique itself could be responsible for the spurious absorption. Fig. 15 shows the results obtained for twenty-four modules from two sets of measurements, first dividing them into four groups of six and then into twelve groups of two. No significant differences between

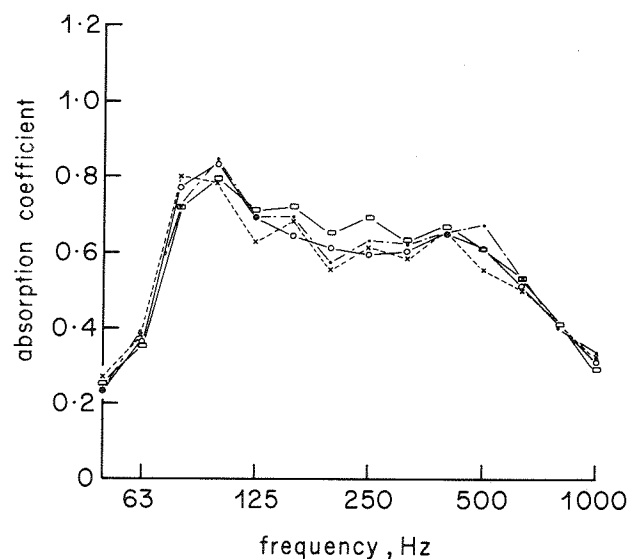


Fig. 16 - Effects of varying the number of diffusing panels in the reverberation chamber.

- — (a) 4 diffusing panels.
- × - - (b) 8 diffusing panels.
- · · (c) 12 diffusing panels.
- — (d) 0 diffusing panels.

the two sets of results are evident.

A second aspect of the measurement technique was the amount of diffusion present in the reverberation room, particularly at low frequencies. The room is normally equipped with twelve diffusion panels hanging from the ceiling. A series of sets of measurements was made with the full quota of twelve diffusers then eight, four and finally no diffusing panels at all. Fig. 16 shows no significant differences were found between the results.

A third aspect of the measurement technique was the method by which the modules were secured to the walls and hence the effect of the space between the rear-panel of the module and the surface of the wall. Normally, the modules are secured to 25 mm x 50 mm battens which are in turn screwed to the walls of the reverberation chamber. This gives approximately 25 mm airspace behind the rear-panel. It was thought that the distance between the front-panel and the wall surface could be significant, which would be true if the rear-panel of the absorber was to some extent transparent to the sound energy because it lacked sufficient stiffness or mass. The distance between the mean position of the mineral-wool layer and the reflecting surface of the wall would be equivalent to a quarter wavelength at about 440 Hz. Such a spaced absorbing layer would give an absorption coefficient peak at the frequency where the spacing was equal to a quarter wavelength. Two sets of comparable measurements were made, one with the modules mounted as usual on the wall and the second with the modules mounted with a 240 mm airspace behind the modules such that their front-panels were 420 mm from the wall. In this case, the frequency at which the spacing would be a quarter wavelength was reduced to 200 Hz. The results for the set of modules spaced well away from the wall should therefore show an increase in the absorption coefficient at 200 Hz and a decrease at 400 Hz when compared with the results for a set of modules mounted in the normal way. Fig. 17 shows these two sets of results in which some of these expected changes are discernable as small variations. However, it is evident from the results shown in Fig. 17 that this is merely a second-order effect and not the primary cause of the spurious absorption at 400 Hz.

4.3 Measurements on modified modular absorbers

4.3.1 Variation of the rear panel

Partly as a result of the evidence obtained from Fig. 17 described in Section 4.2 above, it was

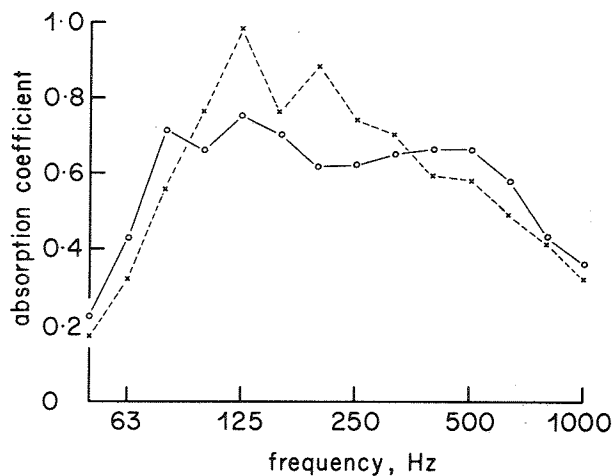


Fig. 17 - Effects of increasing the airspace behind the rear-panel of the module.

○ — (a) normal spacing (25 mm battens)
 x - - - (b) rear-panels spaced at 240 mm from the surface of the wall.

thought that the standard rear-panel made of 3 mm hardboard was unsatisfactory. Therefore, two sets of measurements were made with the standard rear-panel replaced first by 9.5 mm plywood and second by 6.5 mm hardboard. Fig. 18 shows the results, together with the characteristic for the standard module with 3.2 mm hardboard rear-panel. Although some significant differences are apparent, it is evident that an inadequacy of the rear-panel is not the main cause of the spurious absorption at 400 Hz.

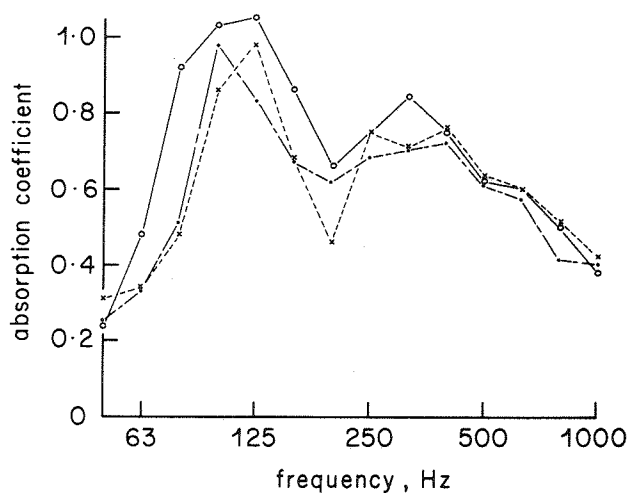


Fig. 18 - Effects of changes in the thickness and material of the rear-panel.

○ — (a) 3.2 mm hardboard.
 x - - - (b) 9.5 mm plywood.
 + · · · (c) 6.5 mm hardboard.

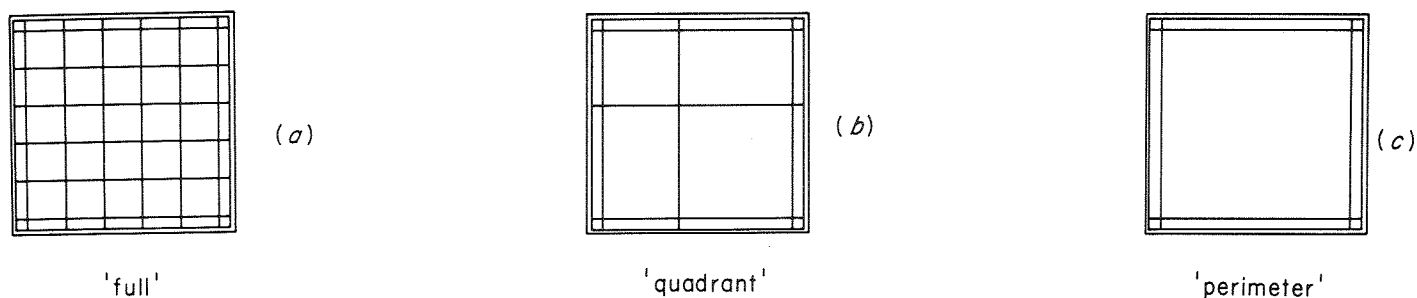


Fig. 19 - Diagrammatic sketches of the three different arrangements of the internal dividers

(a) 'full'

(b) 'quadrant'

(c) 'perimeter'

4.3.2 Variation of the arrangement and the material comprising the internal dividers.

The enclosed space between the rear surface of the mineral-wool layer and the rear-panel is usually divided into about fifty separate compartments by either cardboard or 3 mm hardboard dividers. These serve the dual purposes of preventing lateral interaction between different parts of the absorber for oblique incident sound and supporting the mineral-wool layer against the front-panel. Measurements were made with both hardboard and cardboard dividers, arranged in three different ways. Fig. 19 shows a sketch of these three arrangements called respectively 'full', 'quadrant' and 'perimeter'. Fig. 20 shows the absorption coefficient characteristics for some of the possible combinations. Although some differences are evident, in particular the bandwidth of the low frequency peak in the characteristic is somewhat less with the 'quadrant' or 'perimeter' arrangement, it does not appear

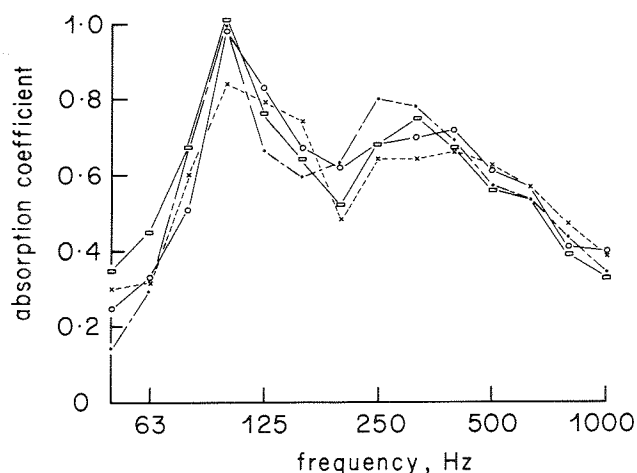


Fig. 20 - Effects of variations in the arrangement and material of the internal dividers.

- ——— ○ (a) 'full' cardboard dividers
- × ——— × (b) 'full' 6.5 mm hardboard dividers
- + ——— + (c) 'quadrant' 6.5 mm hardboard dividers
- ——— □ (d) 'perimeter' 6.5 mm hardboard dividers

that these parameters are closely connected with the unwanted absorption at 400 – 500 Hz. As there is practically no difference between the acoustic performance of the hardboard and cardboard dividers, the choice of cardboard as the preferred material can be made because it is cheaper.

4.3.3 Variation of the material of the mineral-wool layer

Figs. 21 and 22 show the effect of replacing the original mineral-wool layer, 'A' (the original type of material is no longer available in any case), first by a comparable modern material, 'B' and then by a similarly modern but less dense material, 'C', two groups of measurements were made with the normal complete complement of cardboard dividers in the enclosed airspace, Fig. 21, and with the 'quadrant' arrangement of 6.5 mm hardboard

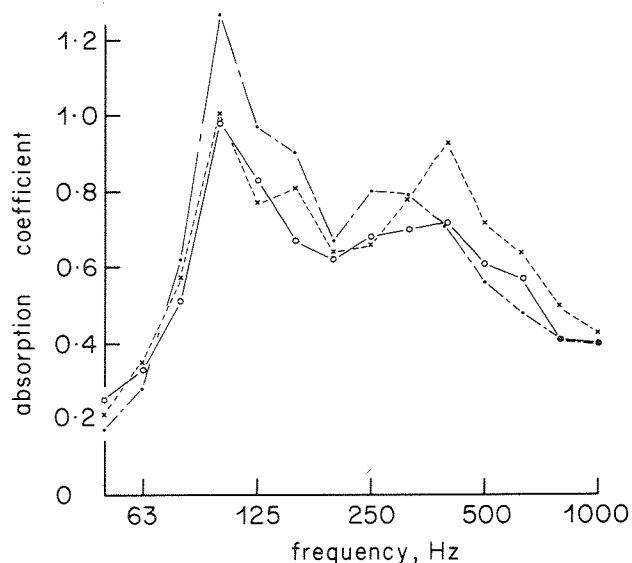


Fig. 21 - Effects of changes in the mineral-wool layer with 'full' arrangement of cardboard dividers.

- ——— ○ (a) mineral-wool, type 'A'
- × ——— × (b) mineral-wool, type 'B'
- + ——— + (c) mineral-wool, type 'C'

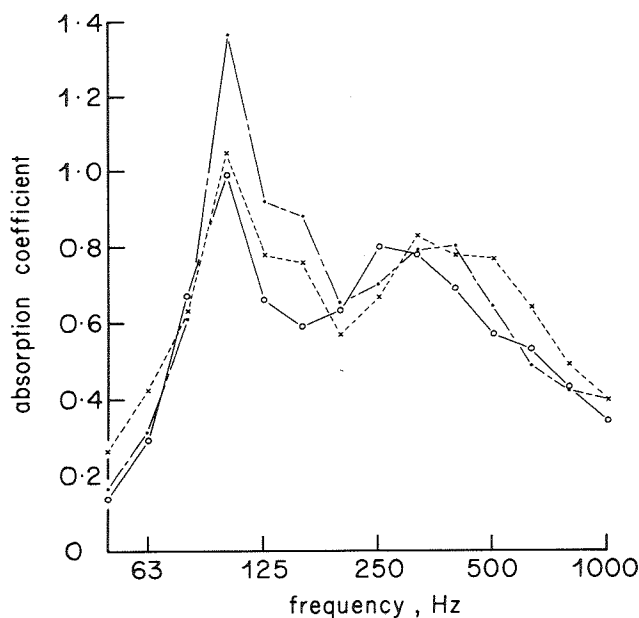


Fig. 22 - Effects of changes in the mineral-wool layer with 'quadrant' arrangement of 6.5 mm hardboard dividers.

- — (a) mineral-wool, type 'A'
- × - - - (b) mineral-wool, type 'B'
- - · - (c) mineral-wool, type 'C'

dividers, Fig. 22. With both arrangements of dividers, the difference in the characteristics around 100 Hz between materials 'A' and 'B' is small but at frequencies above 315 Hz, the absorption is increased with material 'B'. This makes the unwanted absorption relatively worse. However, in both cases, there is a large improvement in the absorption coefficient at 100 Hz using material 'C'.

4.3.4 Investigation of the component parts

After the failure of the above measurements to identify the prime source of the spurious absorption, a series of measurements was carried out in the reverberation chamber on the component parts of the absorber. First, the complete absorbers were measured spaced with their front-panels 450 mm from the wall, to give a reference condition. The rear-panels were then removed leaving the mineral-wool layer and the cardboard dividers in position and a second set of measurements made. Then the entire contents were removed, leaving only the front-panels and the side-frames for a third set of measurements. Finally, the absorbers were removed altogether and the sheets of mineral-wool were hung in their place and a fourth set of measurements made. Fig. 23 shows the results obtained. Without the rear-panel but with the contents of the boxes in their normal positions, the peak in the absorption coefficient at 125 Hz is still (just) evident. This is quite surprising, as the hitherto assumed mechanism of this low-frequency absorp-

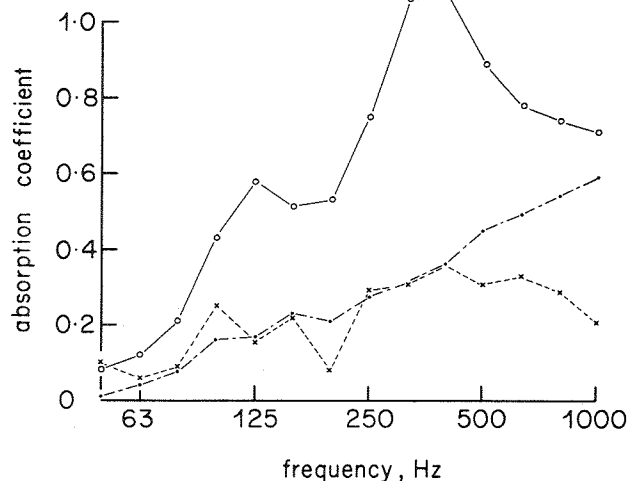


Fig. 23 - Results of measurements on the component parts of the modular absorbers.

- — (a) without rear-panel.
- × - - - (b) front-panel and side-frames only.
- - · - (c) mineral-wool layer only.

tion has been the Helmholtz resonance effect. However, this cannot be present without the rear-panel. The second major feature is the large peak in the absorption coefficient at 315 and 400 Hz for the same condition. This characteristic continues at a relatively high level for all frequencies above 400 Hz as a result of the absorption of the mineral-wool layer which is no longer shielded from the incident high-frequency sound by the rear-panel. So that the effects of the mineral-wool layer could be allowed for, a measurement was also made of the absorption coefficient of the mineral-wool freely suspended in the same relative position within the reverberation chamber as it had occupied when it was inside the modular absorbers.

Subtracting the results for the mineral-wool layer alone from those for the absorbers without their rear-panels it is clear that there is some mechanism of absorption at both 125 Hz and 315 – 400 Hz which can only be due to mechanical vibrations of the box. This hypothesis is reinforced by all the previous experimental results for which no changes were made to the sides of the boxes and also by the fact that the simple theory of Section 3 fails to predict such a high peak in the absorption coefficient, at least at 400 Hz.

4.4 Investigation of panel vibrations

4.4.1 Bracing of front panel

A number of attempts were made to brace the

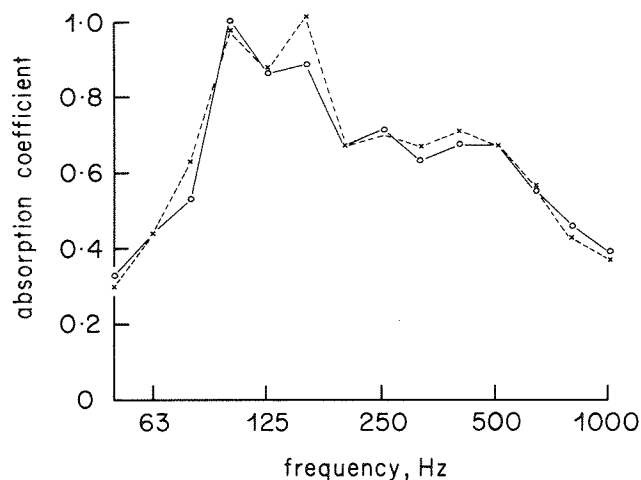


Fig. 24 - Effects of stiffening braces secured to the front-panel.

- — (a) front-panel braced.
- × - - - (b) front-panel unbraced.

front-panels and the side-panels with metal or wooden strips, but these produced no fundamental change in the performance. Fig. 24 shows a comparison between a set of absorbers both with and without a diagonal timber cross-brace of 50 mm x 20 mm section on the front-panel. The only significant difference caused by the brace was a reduction of the absorption coefficient at 160 Hz from 1.06 to 0.88. This elementary and substantially empirical approach was quickly abandoned when it became apparent that all the modes of vibration, and especially those around 400 Hz,

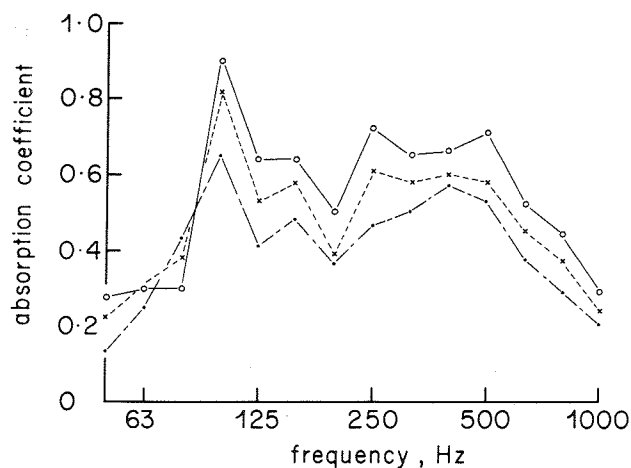


Fig. 25 - Effects of sealing the front-panel perforations on the absorption coefficient characteristic of a standard absorber.

- — (a) standard absorber.
- × - - (b) 1/2 holes sealed.
- + . . . (c) all holes sealed.

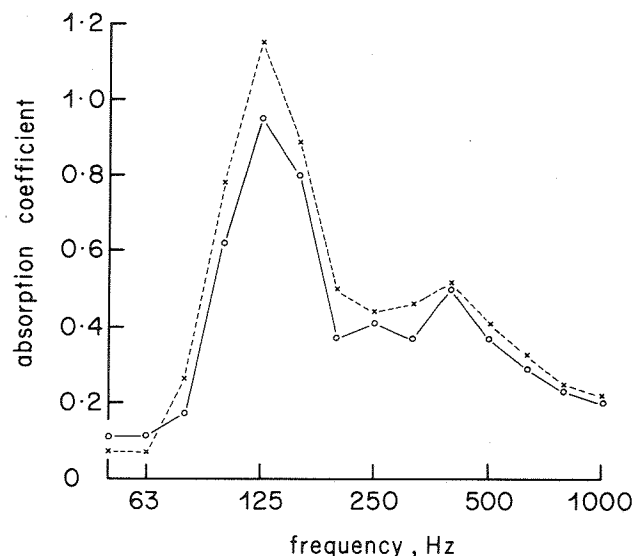


Fig. 26 - Absorption coefficient characteristic of a modular absorber without the mineral-wool layer.

- — (a) with no additional damping
- × - - (b) with a lightweight fabric damping layer.

were high-order modes and were merely being shifted slightly in frequency by the bracing. In addition such bracing does little to alter the losses in the system and it is the loss component which is responsible for the absorption.

4.4.2 Sealing up holes

In order to separate the two possible mechanisms of absorption at about 100 Hz (i.e. the Helmholtz effect and the panel vibrations) a set of measurements was made with some or all of the holes in the front-panel sealed, together with different arrangements of the contents of the absorber. Fig. 25 shows the results obtained with a standard absorber with no holes sealed, one half of the holes sealed and with all of the holes sealed. Apart from a moderate progressive reduction in the absorption coefficient as the holes are sealed up there is no overall change in the absorption characteristic. This shows conclusively that the Helmholtz resonance, if present at all, is so damped by the mineral-wool layer that it has little effect in the standard absorber. This is confirmed by the improved absorption at 125 Hz when the lower density mineral-wool layer was measured (Section 4.3.3 and Fig. 21).

Removing the contents of the boxes gave the results shown in Fig. 26. Compared with the results shown in Fig. 25 the absorption

coefficient at 100 Hz is much greater and at 400 Hz somewhat less than for the standard absorber. By using a layer of very low acoustic resistance fabric (< 1 mks rayl) as a damping layer, instead of the mineral-wool, the absorption coefficient at 100 Hz was improved to greater than 1.1. This shows that, without the mineral-wool layer, the Helmholtz resonance is significant as it is very unlikely that the addition of a very light, open-weave fabric could significantly affect the damping of a panel resonance.

Finally, the effects of progressively sealing up the holes of a set of absorbers with no mineral-wool layer* were investigated. The results are shown in Fig. 27. This shows that the absorption characteristic at about 400 Hz is similar to that measured with the mineral-wool layer in

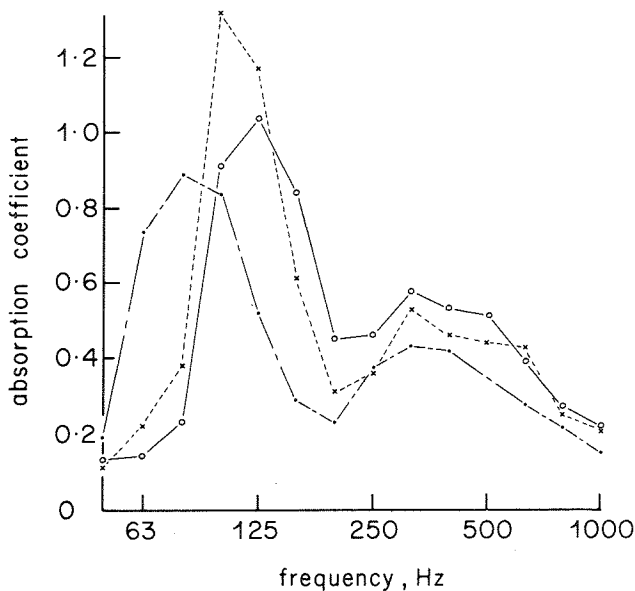


Fig. 27 - Effects of sealing the front-panel perforations on the absorption coefficient of an 'empty' modular.

- — (a) absorber without mineral-wool layer.
- x — (b) as 'a' but with $\frac{1}{2}$ holes sealed.
- • — (c) as 'a' but with all holes sealed.

its normal position at the front of the absorber. It also shows that this absorption is not affected very much by sealing the holes in the front-panel. However, the low-frequency absorption is very significantly affected by sealing the holes. The frequency of the peak in the low-frequency

* Actually, in this case the mineral-wool layer was inside at the back of the boxes adjacent to the rear-panel. This should have very little effect on the 100 Hz panel resonance but much the same effect on the 400 Hz resonance as if it had been in the normal position at the front.

absorption is shifted progressively from 125 Hz to 80 Hz as the number of holes remaining is reduced. This frequency ratio is very nearly $\sqrt{2}:1$ showing that if the compliance of the front-panel itself is neglected, the effective mass of the front-panel is approximately equal to the effective mass of the air in the holes* since both are resonant with the compliance of the enclosed airspace. Separate measurement of the front-panel fundamental resonant frequency (found to be 26 Hz), described below, shows that the inherent acoustic stiffness of the front-panel is approximately 10 times smaller than the stiffness ($=1/\text{compliance}$) of the enclosed airspace.

4.4.3 Investigations of panel vibrations using accelerometers.

The modes of a few resonant frequencies were investigated by systematically moving an accelerometer over the front-panel and drawing contour maps of the amplitude of vibration as a function of position. The absorber was excited into vibration by irradiating it with a sound field. This was a very laborious and time consuming technique which resulted in data which was difficult to interpret. The original basis for this method was to find the optimum position for suitable stiffening braces. However, it became apparent, as stated in Section 4.4.1 above that such a remedy would not work and this approach was abandoned.

4.4.4 Methods using the noise emission of the vibrating panels**

It became clear during the work on panel vibrations that if a panel absorbed sound energy by vibrating then, by reciprocity⁷, it must also radiate sound if caused to vibrate. This offered a simple means by which an individual absorber could be evaluated, at least qualitatively, without requiring the construction of a large number of identical units for testing in the reverberation chamber. This method is not quantitative as it would be almost impossible to measure and reproduce the exact panel vibrations and diffuse radiation pattern at each frequency corresponding to those that exist in the reverberation chamber. However, some useful results were obtained. First, some of the frequencies of mechanical resonance of the modular absorbers were obtained by

* Predicted by C.D. Mathers during early discussions about the theoretical aspects of this work. It was decided not to include this effect in the theoretical model because the model was intended to represent the way in which the absorber had for many years been thought to behave, and because the damping term associated with the mass of the front-panel was unknown.

** Originally suggested by C.D. Mathers.

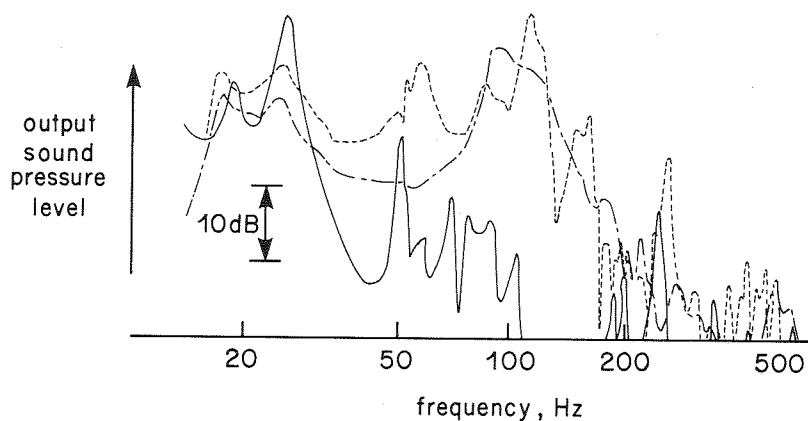


Fig. 28 - Emission response of a single module, excited by an electromagnetic vibrator with a sine-wave drive.

—— (a) front-panel and frame ----- (b) front-panel, frame and rear-panel - . - . (c) standard absorber

attaching an electromagnetic exciter to the box in a free-field room. The exciter was attached in such a way that it had little effect on the resonance (i.e. to a high impedance point). The sound pressure level generated as a result of driving the exciter by a swept frequency tone was detected by a microphone and the results plotted on a chart recorder. Fig. 28 shows a typical set of results from such measurements smoothed to remove the effects of noise and minor perturbations in the response. Curve 'a' in Fig. 28 shows that a simple sheet of 3 mm unperforated hardboard supported at the edges by the wooden side-frame of an absorber box shows a fundamental resonance at 26 Hz, and various modes at higher frequencies. The resonance peaks at lower frequencies are due to the interaction between this primary resonance and the natural frequency of the resonance due to the mass of the entire assembly of panel, side frame and exciter and the compliant suspension of the whole system, which occurred at about 12 – 14 Hz. The fundamental resonant frequency of 26 Hz is the same as that referred to in Section 4.4.2 above.

Curve 'b' of Fig. 28 illustrates the effect of fixing a 6 mm thick rear-panel to the assembly described above. It was in fact a standard A2 absorber with all of the holes sealed up but without the usual contents, i.e. an empty box. Although the resonances at about 26 Hz and 55 Hz are still evident, the most significant feature of curve 'b' is the peak at 112 Hz. This shows that the addition of the compliance of the enclosed airspace has shifted the fundamental resonance of the front panel to a much higher frequency. This curve should be compared with the absorption coefficient

characteristic as measured for the substantially identical condition shown in Fig. 27(c). Finally, Fig. 28 curve 'c' shows a similar plot for a standard A2 absorber. Comparison of curves 'b' and 'c' shows that the damping provided by the mineral-wool layer smooths out the response and significantly reduces the number and amplitude of individual modes.

This method of exciting the absorber into vibration by a swept frequency is not representative of the type of measurements normally made in the reverberation chamber. A more realistic equivalent can be obtained by exciting it by 1/3rd octave, band-limited noise and measuring the acoustic output in each frequency band on a spectrum analyser. This averages the response over relatively wide frequency bands and should give results directly comparable to the results obtained using warble-tone or 1/3rd octave, band-limited noise in the reverberation chamber. In practice, it was found that if a spectrum analyser was used it was far easier to simply excite the absorber with an impulse and thus record all of the frequency bands simultaneously. The electromagnetic exciter and drive amplifier were not necessary for this, a simple sharp blow with a suitable hammer was all that was required. In practice, because the vibrations of the front-panel and the side-panels are closely coupled, it was necessary to adopt a slightly artificial strategy to distinguish between the two resonances. The distribution of the energy of the hammer-blow between the higher and the lower frequencies depended significantly on the characteristics of the hammer-head. It was found that a fairly soft hammer was most suitable for selectively exciting the front-panel resonance and a fairly hard hammer

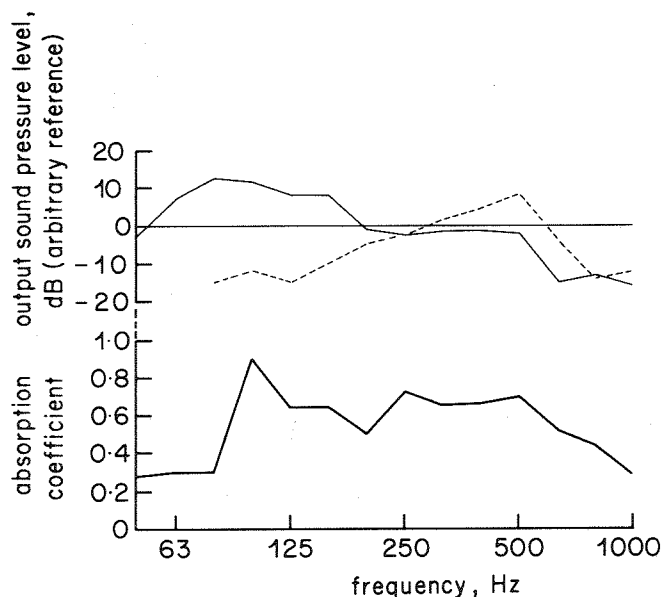


Fig. 29 - 1/3rd octave emission response of a single standard module as a result of an impact, compared with the absorption coefficient characteristic as measured for 24 modules in the reverberation chamber.

— (a) front-panel
 - - - (b) side-panel
 — (c) absorption coefficient characteristic

was most suitable for selectively exciting the side-panel resonance. As in the case for the experiments with the electromagnetic exciter, this method cannot give quantitative results. However, it does have a significant advantage in that it gives a direct visual representation of the panel resonances instantaneously.

The results of one series of measurements made using this method are shown in Figs. 29, 30 and 31. In each case, two sets of results were obtained, one for the front-panel and one for the side-panel. The repeatability of the measurements was found to be better than ± 2 dB after a little practice with the rubber hammer. The figures also include, for reference, the absorption coefficient for the comparable condition of the absorber as measured with a set of twenty-four boxes in the reverberation room. The scales on each of the two parts of Figs. 29, 30 and 31 are in no way comparable, the intention being merely to illustrate the similarity or otherwise of the shapes of the characteristics.

From such comparisons it is evident that there is some clear correlation (especially evident at frequencies above about 250 Hz) between the

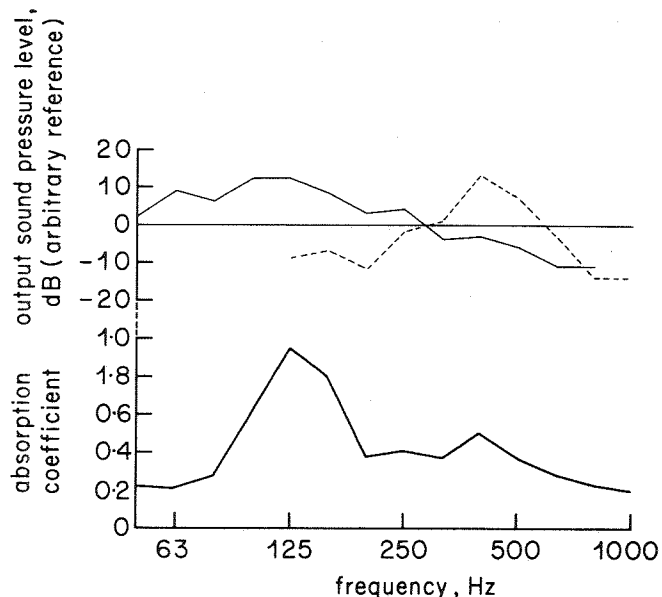


Fig. 30 - 1/3rd octave emission response of a single module with contents but with all the front-panel holes sealed, as a result of an impact, compared with absorption coefficient characteristic as measured for 24 modules in the reverberation chamber.

— (a) front-panel
 - - - (b) side-panel
 — (c) absorption coefficient characteristic

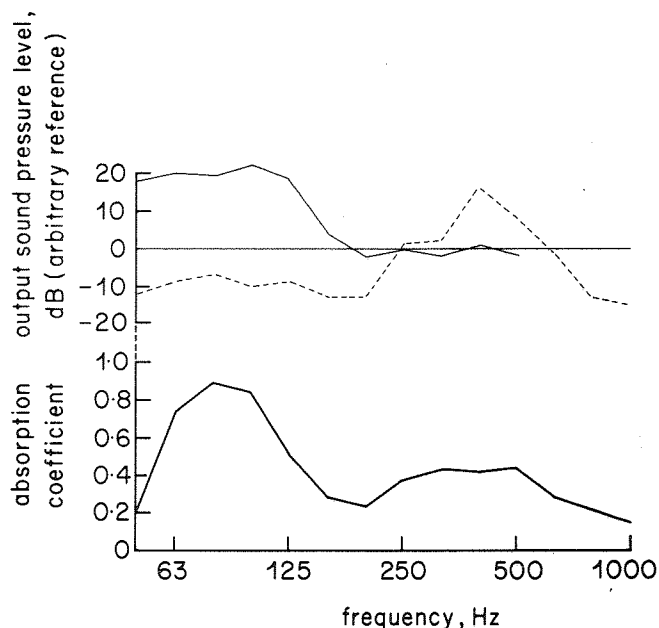


Fig. 31 - 1/3rd octave emission response of a single empty module with all the front-panel holes sealed, as a result of an impact, compared with the absorption coefficient as measured for 24 modules in the reverberation chamber.

— (a) front-panel
 - - - (b) side-panel
 — (c) absorption coefficient characteristic.

measured absorption coefficient and the sound emission characteristics. It was these measurements more than any others which resulted in the conclusion that the excess absorption at or near to 400 Hz was caused by forced vibrations of the side-panels.

4.5. Other confirmatory evidence

A search of the past measurements on other types of low frequency absorber was made. In particular, the type of membrane absorber, mainly used before about 1968, which was constructed in situ from timber studding and was largely without substantial internal divisions equivalent to the sides of the modular absorber, did not show any significant spurious absorption at 400 Hz.

A type of modular absorber, consisting of rectangular metal biscuit boxes with suitable front surfaces also did not show such a significant spurious absorption.

After the completion of the main experimental work described in this report, and initially independent of this investigation, some measurements were made of the performance of a different type of modular absorber (D2). These absorbers were approximately 1.20m x 0.58m x 0.30m deep, that is, the same superficial size as two A2 modules together. They had been constructed incorrectly, using dense mineral wool which was also very rigid, and showed a pronounced spurious absorption at about 250 Hz when measured in the usual way in the reverberation room. The absorption coefficient in the frequency range in which the absorbers were intended to be effective was so low that the spurious absorption was predominant. Measurements were made of the emission spectra of these absorbers when subjected to mechanical impulses. Unlike the A2 modular absorbers, these D2 modular absorbers are not square so that two different sizes of side panel are present, one pair being 0.58m x 0.30m and the other pair being 1.20m x 0.30m. Therefore it could be expected that two different emission spectra would be obtained, depending on which of these two panel sizes was being measured.

Fig. 32 shows the 1/3-octave emission spectra and the absorption coefficient as measured in the reverberation room for these D2 modular absorbers in the same way as Figs. 29, 30 and 31 do for the experimental A2 modular absorbers. The similarity between the shapes of the emission spectra and the absorption coefficient characteristic in the region of 250 Hz are evident, particularly for the smaller panel. That a different type of modular

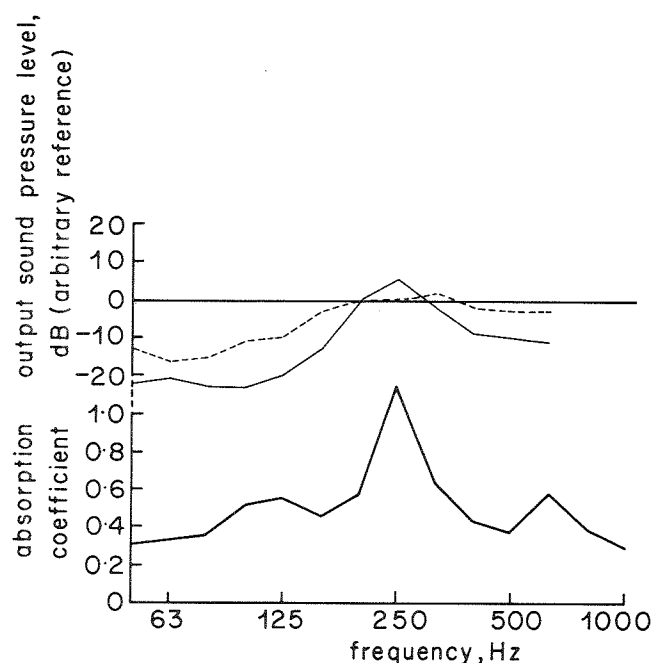


Fig. 32 - 1/3-octave emission response of a single D2 module as a result of an impact, compared with the absorption coefficient as measured for 16 modules in the reverberation chamber

- (a) 0.58m x 0.30m panel
- - - (b) 1.20m x 0.30m panel
- (c) absorption coefficient characteristic

absorber should show a similar correlation between panel resonance emission and absorption coefficient characteristics at a different frequency to that of the A2 is strong evidence that the cause of the spurious absorption has been correctly identified.

5. Principles of the design of a new low frequency modular absorber

From the evidence obtained about the ineffectiveness of the Helmholtz resonance effect and the known narrowness of the working frequency-range of such absorbers, it would seem reasonable to design any new low-frequency absorber to use the vibrating panel or membrane principle. Earlier membrane absorbers which have been used were very effective when working but, the bonded-hardboard and roofing-felt units in particular, lost their effectiveness either because the glue joint between the two elements of the front-panel was unsatisfactory from the beginning or as a result of the degradation of the glue joint with time. Those units which used roofing-felt alone as the front element tended to dry out and become inflexible and, in any case, would not be suitable for new applications because of the fire-risk inherent in such a

material. It may be that, with modern adhesives, a satisfactory front-panel could be designed using a sheet-metal and a synthetic damping material (such as a plasticised polyvinyl butyral resin*) multilayer combination.

Whatever form a new absorber takes, it is evident that great care must be taken in the design to ensure that any panel resonances, except the front-panel fundamental, must be either designed to lie well outside the audio-frequency range or be so heavily damped that they cannot absorb a significant amount of energy. Alternatively, it may be possible to design the side-panel resonances to be at the same or near to the resonant frequency of the front-panel. In this case great care would have to be taken to ensure that there was no detrimental interaction between the two resonances.

6. Consequential implications for other types of absorber

It was earlier stated that the A3 type of wide-band absorber should, theoretically be ineffective at frequencies lower than about 400 Hz. In practice, the A3 type of absorber is effective down to frequencies as low as about 200 Hz. It is now clear that the same kind of panel resonances are responsible for this extension of the working range. In the case of the highly perforated front-panel of the A3, the effects of the perforations may well be sufficient to give a resonant frequency for the front-panel of about 200 Hz (although a simple calculation based solely on the mass difference shows that the ratio of the resonant frequency with 20% perforations should be about $1/\sqrt{0.8}$ that is, about 1.11 times the resonant frequency of the unperforated panel). However, a full study of the behaviour of the A3 is outside the scope of this work and would require a substantial amount of further work to be quantified more accurately.

7. Conclusions

A study of the acoustic behaviour of a low-frequency modular absorber has been carried out. The difficulty of this work was greatly increased by the chance coincidence of a number of effects, for example, the coincidences between the side-panel resonant frequency and the frequency at

which the spacing between the mineral-wool layer and the rear-panel is one quarter wavelength, and between the fundamental resonant frequency of the front-panel and the Helmholtz frequency.

Although the unwanted absorption at about 400 Hz was evident in the initial measurements on the prototype units, it was not considered at that time to be sufficiently important to be investigated thoroughly. Since then some change has occurred which makes this unwanted absorption somewhat worse than it was originally. It is now sufficiently significant to cause problems when the A2 and A3 modular absorbers are used together for the acoustic treatment of studios and control rooms (cubicles).

As a result of the experimental and theoretical work recorded in this report, it is evident that the original basis of the design of the A2 low-frequency absorber was not sufficiently understood and that the mechanism of operation of these units is not that on which the design was based.

For the frequency region in which the absorption coefficient is required to have a large value, that is, around 100 Hz, the supposed resonance of the mass of the air contained within the holes of the perforated front-panel with the compliance of the volume of air enclosed by the box (the Helmholtz effect) is so heavily damped by the thick layer of mineral-wool that it is effectively non-existent. Instead, the actual sound absorption in this range of frequencies is provided by the resonant system formed by the mass of the front-panel vibrating as a limp membrane, and the compliance of the volume of air enclosed within the box. This resonance is damped partially by the inherent damping of the hardboard panel and partially by its' substantially uncontrolled contact with the mineral-wool layer. That this mode is accidental is indicated by the fact that if it had been deliberate then the front-panel would not have to be perforated, with a consequent saving in manufacturing costs.

If the mineral-wool layer is omitted the Helmholtz effect becomes significant. The combination of this in parallel with the front-panel resonance raises the frequency at which the absorption coefficient becomes a maximum when compared with the resonant frequency of the front-panel alone. This could be used to give two different low frequency absorbers, one with an unperforated front-panel and a peak absorption at about 80 Hz and the second with a perforated front-panel and a peak absorption at about 125 Hz. However, relying on the inherent damping of the

* This material is virtually unique in that it has a low transition frequency at room temperature. This results in a high dynamic damping factor at low frequencies and room temperature. It is, or can be made to be, extremely adhesive although the time dependence of this adhesion property is unknown at present.

front-panel material is not a satisfactory solution without knowledge of the consistency of production and the long-term stability of this property of the material. In any case, it is very unlikely that the damping coefficient provided in this way would be the correct value to give the maximum sound absorption coefficient, although in the experiments carried out during the course of this work, high values for the absorption coefficient (>1.0) were accidentally obtained by this means.

The region of unwanted absorption around 400 Hz has been found to be caused by the damped vibrations of the 9 mm thick plywood side-panels. This is a feature of the design which, as far as can be determined, has remained unaltered since the production of the prototypes. It is therefore surprising that any change in the absorption coefficient characteristic has occurred in the region of 400 Hz. However it may be that the detail of the material currently used and hence the inherent internal damping is different to that used originally. For example 9 mm resin-bonded 7-ply birch plywood has a Youngs Modulus approximately 10% higher and a damping coefficient about 13% lower than the comparable Casein-bonded plywood.

It may be possible to design a new low-frequency modular absorber based on the existing A2 module. This may have an unperforated front-panel if a suitable combination of superficial mass and damping coefficient can be achieved. Alternatively it may be perforated and use a combination of panel resonance and Helmholtz resonance to achieve the required absorption characteristic.

In the design of a new low-frequency absorber, two aspects of the design will require special consideration. The first is the long-term stability of the characteristics of the front-panel, particularly if the damping is inherent in or combined with the structural element and the second is concerned with the resonant modes of the supporting structure and the associated damping terms.

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Appendix 1 - Summary of Modular Absorber Types — (Based on the 'Guide to Acoustic Practice' issued by the BBC Architectural and Civil Engineering Department)

Modular absorbers are identified by a two-part designation consisting of a single letter and a number, for example, 'A2'.

The letter denotes the overall depth of the unit (from front to back) and consists of:

A	183.5 mm	} all with 30 mm thick mineral-wool
B	107.5 mm	
C	215 mm	

The number denotes the internal arrangement, construction and the material comprising the front-panel:

1. 12.5 mm square, galvanised weldmesh in front of the mineral-wool, with an approved, open-weave fabric covering the mineral-wool.
2. 0.5% open-area perforated hardboard (generally 3 mm diameter holes on 38 mm centres)

over the mineral-wool.

3. 20% open-area perforated hardboard (generally 3 mm diameter holes at 6.5 mm centres) over the mineral-wool.
4. 12.5 mm "chicken-wire" over the mineral-wool (used occasionally, behind fixed curtaining).
5. Approved open-weave stretched fabric over the mineral-wool.

The construction of the type 'C' modular absorber differs from types 'A' and 'B' in that the mineral-wool layer is 50 mm thick and, in the type 'C2' the perforated hardboard is positioned behind the mineral-wool layer, which is located at the front by 12.5 mm square galvanised weldmesh.

Ocasionally, specially designed modular absorbers have numbers not included in the above list.

Appendix 2

Reactive sources and loads

Consider the circuit shown in Fig. A1

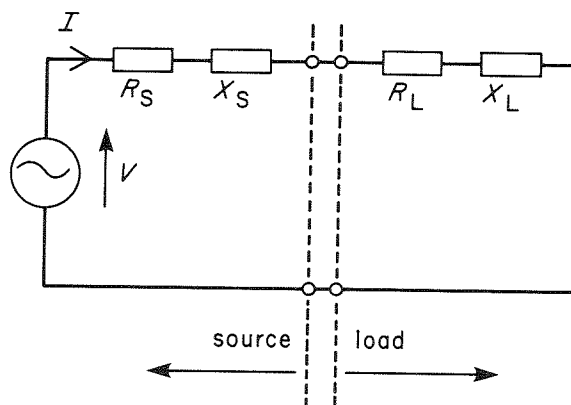


Fig. A1 - Circuit of reactive source and load impedances

$$\text{RMS current, } I = V / ((R_S + R_L)^2 + (X_S + X_L)^2)^{1/2}$$

Power dissipated in the load, P_1 ;

$$\begin{aligned} P_1 &= I^2 R_L \\ &= V^2 R_L / (R^2 + X^2) \end{aligned}$$

where

$$\begin{aligned} R &= R_S + R_L \\ X &= X_S + X_L \end{aligned}$$

and $(R^2 + X^2) = (\text{total circuit impedance})^2$

Power, P_2 , which could be absorbed by a load whose impedance is the complex conjugate of the source, that is:

$$R_L + jX_L = R_S - jX_S$$

is given by:

$$P_2 = V^2 / (4R_S)$$

The ratio of these powers P_1/P_2 is given by:

$$P_1/P_2 = 4R_S R_L / (R^2 + X^2)$$

Replacing these general terms by the symbols used in Section 3 gives:

$$\alpha = P_1/P_2 = 4R_R (R_A + R_B) / (R^2 + X^2)$$

CJS

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